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USAAVLABS TECHNICAL REPORT 70-63

EVALUATION OF GEOMETRY FACTORS IN ROLLER BEARINGS

By

Vernon M. Zwicker

November 1970

U. S. ARMY AVIATION MATERIEL LABORATORIES  
FORT EUSTIS, VIRGINIA

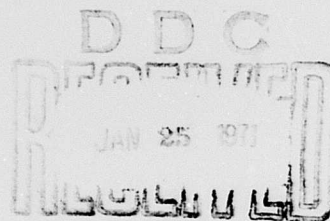
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**DEPARTMENT OF THE ARMY**  
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**FORT EUSTIS, VIRGINIA 23604**

The objective of this program was to determine, through design experimentation and statistical analyses, the effect of geometry variables in relation to roller skidding and end wear in cylindrical roller bearings.

This report presents the results of this investigation. The data derived from this program will help to advance the state of the art in roller bearing technology.

This command concurs in the conclusions and recommendations made by the contractor.

Task 1G162203D14414  
Contract DAAJ02-68-C-0107  
USAAVLABS Technical Report 70-63  
November 1970

EVALUATION OF GEOMETRY FACTORS IN ROLLER BEARINGS

Final Report

EDR 6772

By

Vernon M. Zwicker

Prepared by

Detroit Diesel Allison Division • General Motors  
Indianapolis, Indiana

for

U.S. ARMY AVIATION MATERIEL LABORATORIES  
FORT EUSTIS, VIRGINIA

This document is subject to special export controls, and each transmittal to foreign governments or foreign nationals may be made only with prior approval of U. S. Army Aviation Materiel Laboratories, Fort Eustis, Virginia 23604.

## SUMMARY

A four-stage test rig and related systems with recirculating lubrication were designed and built for testing 40-millimeter, cylindrical roller bearings with specific two-level values of five geometry factors and two operational factors to determine which factors or combination of factors affects roller end wear and skidding. The program was based on a half-replicate of a  $2^7$  factorial experiment. Each bearing was scheduled to run five 2-hour cycles consisting of alternate 10-lb, and 400-lb radial loads and speed accelerations and decelerations between 25,000 and 35,000 rpm. After testing, the bearings were carefully examined and measured for evaluation of the test variables with respect to skidding and roller end wear. Test results were analyzed using a stepwise regression program operated with the Allison computer facilities.

Operation under misalignment, once considered to be extremely injurious, was feasible, with the proper combination of variable levels, even to the extent of preventing roller end wear. The variable of greatest significance was roller end clearance, always requiring the low level value. Roller pocket squareness was next in order of importance, indicating that close control was required. Other significant variables were crown and internal clearance. The other variables of flange angle and plane of misalignment were not significant.

## FOREWORD

This is the final report on the project entitled "Evaluation of Bearing Geometry Factor." This project was conducted during the period July 1, 1968 to July 15, 1970, for the U.S. Army Aviation Materiel Laboratories (USAAVLABS) under Contract DAAJ02-68-C-0107, Task 1G162203D14414.

USAAVLABS technical direction was provided by Mr. R. Givens.

The physical testing of the bearing specimens was performed by Midwest Aero Industries Corporation, Royal Oak, Michigan.

Statistical analysis of roller end wear test data was prepared by R. L. Finch, Statistical Services Section of the Reliability Department.

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## INTRODUCTION

The purpose of this project was to determine the factors or combination of factors that affects skidding and roller end wear in cylindrical roller bearings. These two types of damage have long been a problem with roller bearings. A statistically designed experiment consisting of five geometric and two operational factors was tested in a half-replicate of a  $2^7$  factorial experiment under varying operating conditions of load and speed.

## BACKGROUND

There are numerous failure modes in the broad field of cylindrical roller bearing operation in aircraft engines and gearboxes. They cross the full spectrum from catastrophic failures that suddenly disable an engine to the slow, gradual deterioration that can be tolerated safely until the bearing is removed at the next overhaul. Whenever the former type occurs, vigorous action is taken to define and correct the problem. Consequently, less critical modes often do not receive their deserved attention. Roller end wear and skidding are typically in this category. Both conditions can, and frequently do, deteriorate to a critical state where a catastrophic failure may occur. This program was initiated to comprehend and find solutions to problems in this class and to generally advance the state of the art.

## ANALYSIS OF EFFECT OF DESIGN AND OPERATIONAL FACTORS

A great deal of consideration was given to selecting the seven parameters. Misalignment is deemed to be the strongest reason for end wear. When the roller passes through the region normal to the axis of misalignment, it experiences its greatest concentration of load along a small percentage of its length at one end. This loading imposes a strong tendency to turn the roller from its plane of rotation which must be resisted by the guiding effect of the two flanges. This tends to produce roller end wear.

Closely associated with the effect of misalignment is the roller end clearance. The greater the clearance, the farther the roller is able to divert from its true course; consequently, the faster the ends will wear. The concentration of forces required to guide the rollers increases as the end clearance and skew angle increase; therefore, the lubricant becomes less effective.

The diametral clearance also becomes a factor, perhaps to a lesser extent with respect to roller end wear than to roller skidding. Large clearance reduces the number of rollers carrying the load and providing traction to drive the roller and cage assembly, which normally has considerable drag. When the rollers that drive can no longer overcome the drag, skidding takes place. Under high speed and light load, this phenomenon is quite common. With respect to roller end wear, the effect of internal clearance may be much less. A roller seated in a concave surface (the outer race) will tend to conform to the concavity. However, any force great enough to overcome this tendency requires that the roller take on a slightly skewed position and, in so doing, moves a very small distance toward the center of curvature (axis of the outer ring) astride the outer ring roller pathway, becoming totally unsupported in the center. A large internal clearance permits the movement toward the center of curvature, the skewing, and the consequent roller end wear.

The layback angle of the guide flanges affects the lubrication of the roller ends by providing a little more area and a more favorable condition of convergence for the lubricant.

The skewed cage pockets apply a force on the cylindrical surface of the rollers near the end that first makes contact with the cage pocket cross-bars, forcing a turning tendency which must be resisted by the guide flanges to the flat ends of the rollers.

The roller crown indirectly affects end wear, in that a roller carrying its load primarily in the center portion rather than along its full length will skew much easier under the influence of other factors.

The final parameter chosen (because of its possible influence on roller end wear and skidding) is the plane of misalignment with respect to the load vector. The two levels represent rollers entering the load zone under maximum misalignment and leaving the load zone under maximum misalignment. Rollers generally behave differently upon entering the load zone than upon leaving, even under well-aligned conditions.

In general, any condition or combination of conditions that promotes roller end wear increases the drag on the roller-cage assembly, and in turn, any increase in drag promotes the tendency toward skidding.

## STATEMENT OF THE PROBLEM

The rolling elements of a cylindrical roller bearing are guided between two flanges integral with either the inner ring or the outer ring. Frequently, the ring without the two flanges has one flange that is used for axial location. In this discussion, only guiding will be considered. The relative motion between the inside face of a flange and the end of a roller is intermittent sliding; i.e., any point on either of these surfaces is not subjected continuously to sliding. The area in contact at any instant is relatively small. For these reasons, and because the application of lubricant, particularly to the inner ring, is difficult, roller end wear has been a perpetual problem in roller bearings without any practical solution. The lubrication film will not support the load as the roller tends to skew, resulting in metal-to-metal contact of the roller and ring flange. Maximum wear occurs on the roller at the point of tangency of the roller corner radius and its flat end. In some cases, the wear progresses far enough to permit the roller axis to rotate  $90^\circ$  and to align itself in the plane of bearing rotation.

Another common roller bearing problem is sliding or skidding of the rollers on the roller tracks, particularly on the inner race. This type of motion causes breakdown of the oil film, resulting in two kinds of damage:

- A surface having a dull, "frosty" appearance
- A surface having a glossy, polished appearance

The two differ only in rate of slip, the former resulting from a high rate and the latter from a lesser rate. Most rolling element bearings exhibit a slight, degree of slip, even under the most favorable environmental conditions.

In general, neither roller end wear nor skidding results in catastrophic failures, but they produce a slow and gradual deterioration requiring early replacement. Although some design modifications have been applied to prevent or retard the damage, no known comprehensive effort has been made to better understand the causes.

## APPROACH TO THE PROBLEM

An opinion survey was conducted among a number of individuals and companies to form a basis of investigating end wear. The following seven factors emerged as those most likely to affect end wear:

- Angle of misalignment
- Plane of misalignment with respect to load vector
- Degree of crown
- Roller end clearance in guide flanges
- Internal clearance
- Flange angles (layback)
- Cage pocket skew

Operating conditions, viz., alternating high and low loads and speed accelerations and decelerations, were introduced to include the investigation of skidding in the same program.

The best approach to a problem with so many variables is by the well-known design experiment in which each variable is analyzed in light of all the others. Normally, 128 specimens would be required for a  $2^7$  factorial experiment, but, in the interest of economy, a half-replicate consisting of 64 specimens was chosen. Twenty-eight additional bearings were purchased. Four were required for the original assembly and running in the test rig; four, to establish the running conditions and test time; and four, to verify the results of the first test. The final 16 were required for generating data used to define curvilinear response to establish test repeatability of randomly selected tests in the 16 sets (64 bearings) of the Phase I schedule, and for repeating certain nonqualifying tests among the 64 bearings.

The specimen chosen for this program was a conventional 40 mm × 62 mm × 12 mm cylindrical roller bearing having a two-flanged outer ring, a nonflanged inner ring, 18-6 mm diameter by 7.5 mm long rollers, and a one-piece iron-silicon bronze, unplated cage (Figure 1). All bearings were made from the same heat of degassed AISI 52100 steel to AFBMA grade 5 tolerances. Dimensional stabilization was standard for this size bearing, viz., 338°-347°F for the inner ring, 454°-464°F single stabilized for the outer ring, and 455°-464°F double stabilized for the rollers. The difference in inner and outer ring stabilizing temperatures is based on the ring diameter. Test factors 1 through 7 (Figure 2), were evaluated at high and low levels, as shown in Table I.

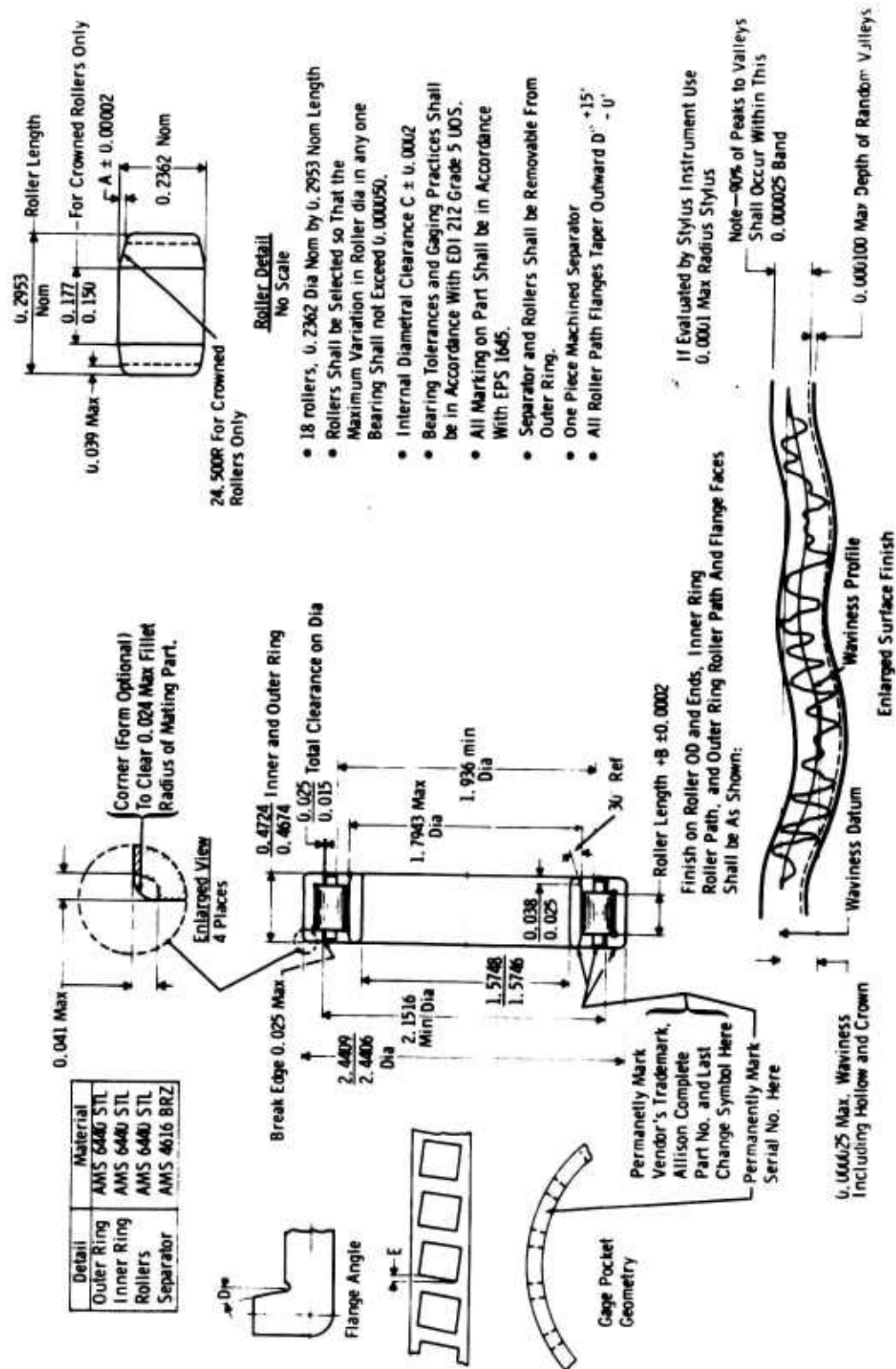


Figure 1. Test Bearing.

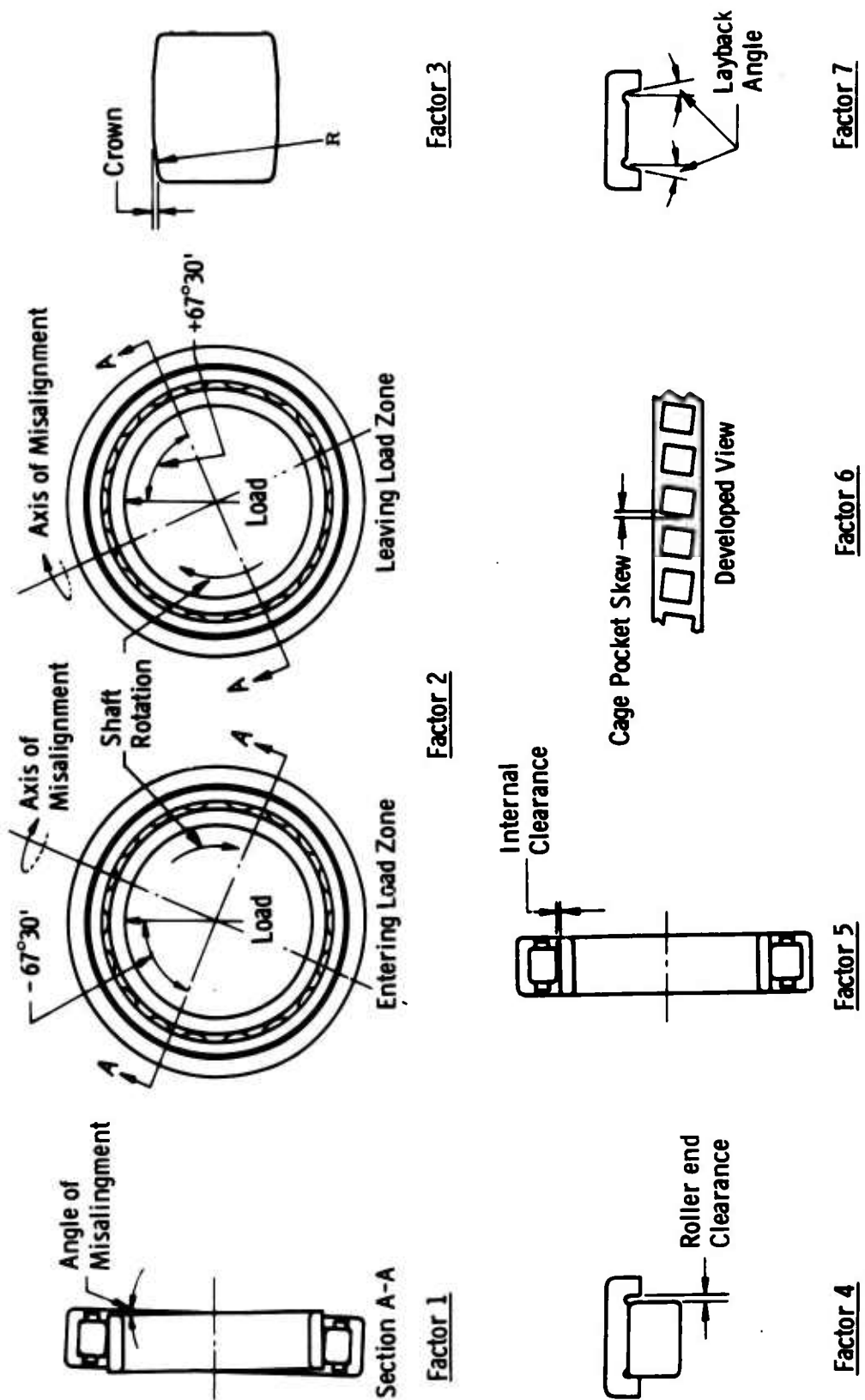


Figure 2. Test Factors.

TABLE I. TEST FACTORS EVALUATED		
Factor	LOW LEVEL	HIGH LEVEL
1* Angle of misalignment	0 degrees	0 degrees, 10 minutes
2* Plane of misalignment with respect to load vector	-67 degrees, 30 minutes	+67 degrees, 30 minutes
3 Degree of crown (in.)	0	0.0001
4 Roller end clearance in guide flanges (in.)	0.0007±0.0002	0.0030±0.0002
5 Internal clearance (in.)	0.0007±0.0002	0.0022±0.0002
6 Flange angle (layback)	0° +15' - 0	0°45' +15' -0
7 Cage pocket skew (in.)	0	0.005
*Installation factors		

Table II lists 79 combinations of the seven factors applicable to each bearing; they are identified by part number and serial number. Bearing part number EX92367-2, serial number GG0068, was severely damaged during build assembly and could not be tested. Therefore, one of the 16 bearings (Serial No. GG0078) in Phase II of the program was made with the same values of the factors and tested to provide data for that cell in the test array. This accounts for 79 instead of 80 combinations.

TABLE II. TEST BEARING FACTOR COMBINATIONS								
P/N EX-92367-XX	S/N GG00XX	Degree Misalignment (min)	Plane of Misalignment (deg)	A Crown (in.)	B Roller End Clearance (in.)	C Internal Clearance (in.)	D Flange Angle (min)	E Pocket Skew (in.)
1	41	10	-67	0	0.0047	0.0007	0	0
1	37	0	---	0	0.0007	0.0007	0	0
2	76	0	---	0.0001	0.0007	0.0007	0	0
2	68 (78)	10	+67	0.0001	0.0007	0.0007	0	0
3	32	0	---	0	0.0030	0.0007	0	0
3	34	10	+67	0	0.0030	0.0007	0	0
4	63	10	-67	0.0001	0.0030	0.0007	0	0
4	55	0	---	0.0001	0.0030	0.0007	0	0
5	23	0	---	0	0.0007	0.0022	0	0
5	16	10	+67	0	0.0007	0.0022	0	0
6	47	10	-67	0.0001	0.0007	0.0022	0	0
6	67	0	---	0.0001	0.0007	0.0022	0	0
7	44	10	-67	0	0.0030	0.0022	0	0
7	20	0	---	0	0.0030	0.0022	0	0
8	49	0	---	0.0001	0.0030	0.0022	0	0
8	56	10	+67	0.0001	0.0030	0.0022	0	0
9	45	10	-67	0	0.0007	0.0007	45	0
9	21	0	---	0	0.0007	0.0007	45	0
10	73	10	-67	0.0001	0.0007	0.0007	45	0
10	52	0	---	0.0001	0.0007	0.0007	45	0
11	26	10	-67	0	0.0030	0.0007	45	0
11	28	0	---	0	0.0030	0.0007	45	0
12	57	0	---	0.0001	0.0030	0.0007	45	0
12	58	10	+67	0.0001	0.0030	0.0007	45	0
13	14	10	-67	0	0.0007	0.0022	45	0
13	42	0	---	0	0.0007	0.0022	45	0
14	46	0	---	0.0001	0.0007	0.0022	45	0
14	69	10	+67	0.0001	0.0007	0.0022	45	0
15	15	0	---	0	0.0030	0.0022	45	0
15	27	10	+67	0	0.0030	0.0022	45	0
16	62	10	-67	0.0001	0.0030	0.0022	45	0
16	60	0	---	0.0001	0.0030	0.0022	45	0
17	30	0	---	0	0.0007	0.0007	0	0.005
17	31	10	+67	0	0.0007	0.0007	0	0.005
18	66	10	-67	0.0001	0.0007	0.0007	0	0.005
18	74	0	---	0.0001	0.0007	0.0007	0	0.005
19	29	10	-67	0	0.0030	0.0007	0	0.005
19	25	0	---	0	0.0030	0.0007	0	0.005
20	54	0	---	0.0001	0.0030	0.0007	0	0.005
20	48	10	+67	0.0001	0.0030	0.0007	0	0.005
21	19	0	---	0	0.0007	0.0022	0	0.005
21	18	10	+67	0	0.0007	0.0022	0	0.005
22	51	0	---	0.0001	0.0007	0.0022	0	0.005
22	70	10	+67	0.0001	0.0007	0.0022	0	0.005

TABLE II. -Continued								
P / N EX92367-XX	S / N GG00XX	Degree Misalignment (min)	Plane of Misalignment (deg)	A Crown (in.)	B Roller End Clearance (in.)	C Internal Clearance (in.)	D Flange Angle (min)	Pocket Skew (in.)
23	22	0	---	0	0.0030	0.0022	0	0.005
23	17	10	+67	0	0.0030	0.0022	0	0.005
24	53	10	-67	0.0001	0.0030	0.0022	0	0.005
24	50	10	---	0.0001	0.0030	0.0022	0	0.005
25	33	10	-67	0	0.0007	0.0007	45	0.005
25	36	0	---	0	0.0007	0.0007	45	0.005
26	64	0	---	0.0001	0.0007	0.0007	45	0.005
26	75	10	+67	0.0001	0.0007	0.0007	45	0.005
27	24	0	---	0	0.0030	0.0007	45	0.005
27	43	10	+67	0	0.0030	0.0007	45	0.005
28	61	10	-67	0.0001	0.0030	0.0007	45	0.005
28	59	0	---	0.0001	0.0030	0.0007	45	0.005
29	38	0	---	0	0.0007	0.0022	45	0.005
29	40	10	+67	0	0.0007	0.0022	45	0.005
30	71	10	-67	0.0001	0.0007	0.0022	45	0
30	72	0	---	0.0001	0.0007	0.0022	45	0.005
31	39	10	-67	0	0.0030	0.0022	45	0.005
31	35	0	---	0	0.0030	0.0022	45	0.005
32	65	0	---	0.0001	0.0030	0.0022	45	0.005
32	77	10	+67	0.0001	0.0030	0.0022	45	0.005
33	79	0	---	0	0.0007	0.0010	0	0
34	80	0	---	0	0.0030	0.0025	0	0.005
35	81	0	---	0	0.00185	0.00175	0	0.0025
36	82	0	---	0.0001	0.0007	0.0010	27.5	0.005
37	83	0	---	0.0001	0.00185	0.00175	27.5	0
38	84	0	---	0.0001	0.0030	0.0025	27.5	0.0025
39	85	0	---	0.00005	0.0007	0.00175	27.5	0.0025
40	86	0	---	0.00005	0.00185	0.0025	27.5	0.003
41	87	0	---	0.00005	0.0030	0.0010	27.5	0
42	88	10	+67	0	0.0007	0.0025	27.5	0
43	89	10	-67	0	0.00185	0.0010	27.5	0.0025
44	90	10	+67	0	0.0030	0.00175	27.5	0.005
45	91	10	-67	0.0001	0.0007	0.0025	45	0.005
46	92	10	+67	0.0001	0.00185	0.0010	45	0
47	93	10	-67	0.0001	0.0030	0.00175	45	0.0025

All bearings except the four required for original rig buildup were furnished by SKF Industries, Inc., Philadelphia, Pennsylvania, with the following measurements recorded and furnished for each bearing:

- Inner ring bore diameter
- Inner ring roller path eccentricity
- Inner ring roller path taper
- Inner ring roller path waviness
- Inner ring roller path finish
- Outer ring outside diameter
- Outer ring roller path eccentricity
- Outer ring roller path waviness
- Outer ring roller path taper
- Outer ring roller path finish
- Outer ring flange finish
- Flange angle (stamped side)
- Flange angle (nonstamped side)
- Roller end clearance
- Radial internal clearance
- Roller finish (outside diameter)
- Roller finish (ends)
- Roller crown
- Cage pocket squareness
- Roller pocket clearance (end)
- Roller pocket clearance (circumferential)

## TEST EQUIPMENT

The test rig, shown in Figures 3, 4, and 5, has four stations in which four bearings were tested simultaneously; two bearings (positions 1 and 4) were supported in a stationary housing, while the other two bearings (positions 2 and 3) were in movable housings that provided the radial load through a hydraulic piston (Figure 5). Position numbers are from left to right. The two independent housings were connected to a single piston rod through flexible joints to assure equal load distribution between the two bearings.

The test bearings, having no flanges on the inner rings, did not axially restrain the shaft. Furthermore, when cylindrical roller bearings run misaligned, as half of these did, an axial load, or thrust, is inherently imposed on the shaft. In order to position the shaft against these forces, and from any similar effects of the drive coupling, a duplex pair of ball bearings was provided in the stationary housing. Sufficient radial clearance in the housing ensured that the ball bearings did not share radial load intended for the test bearings.

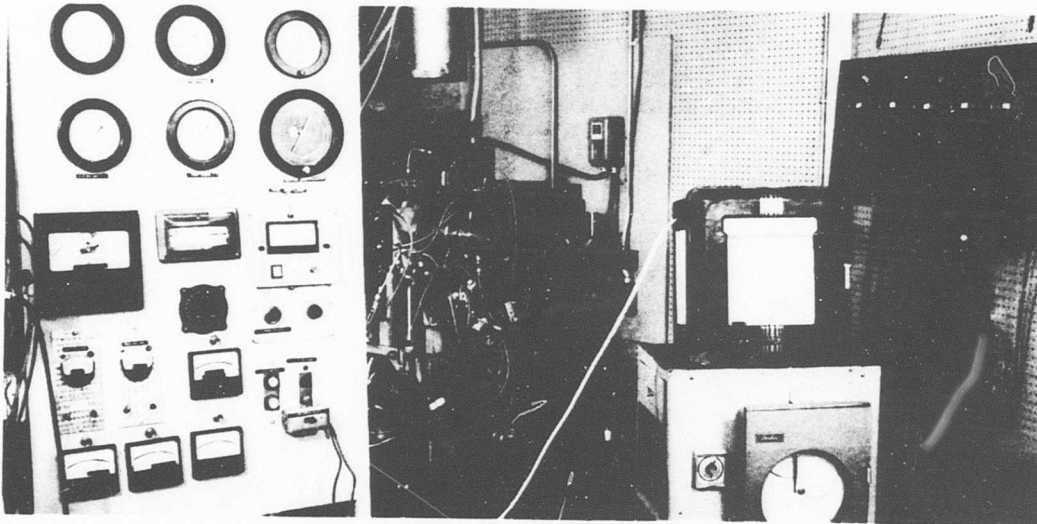


Figure 3. High-Speed Roller Bearing Installed in Test Rig.

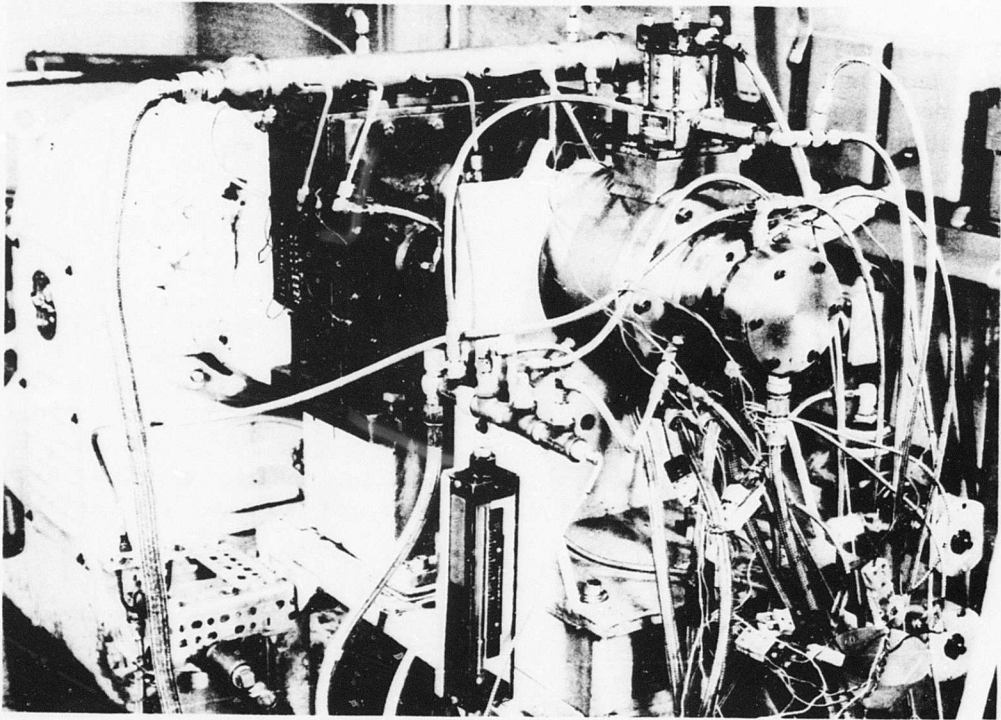


Figure 4. Close-up of High-Speed Roller Bearing Rig Ready for Test.

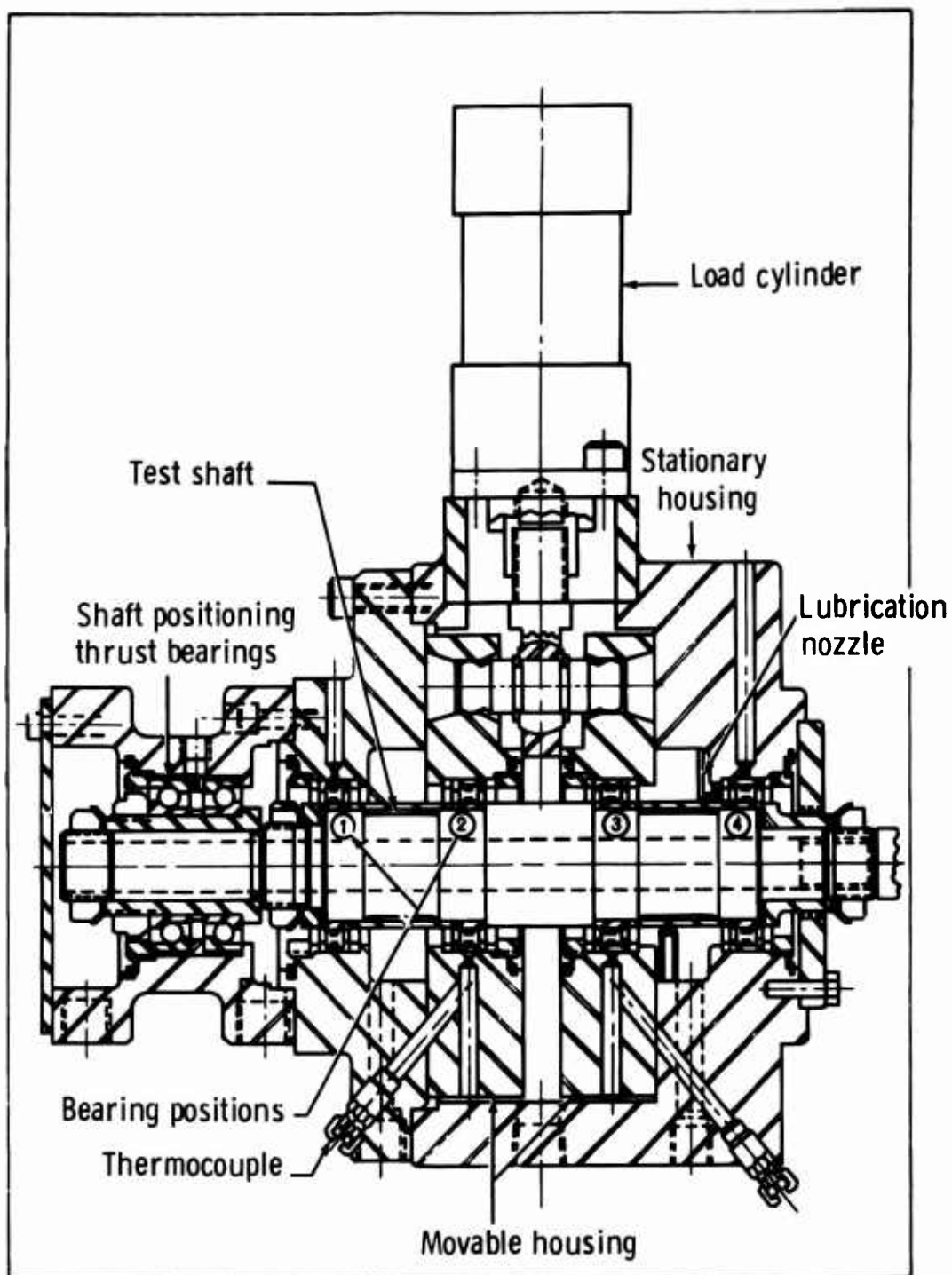


Figure 5. Test Rig Cross Section.

The test bearings were lubricated by a single 0.032-inch diameter jet directed at the space between the inner ring outside diameter and the roller cage inside diameter and located 130 degrees ahead of the load zone. In order that the environment of all four test bearings be as nearly alike as possible, the oil jets provided for the two movable bearings were displaced 180 degrees from those for the bearings in the stationary positions. Two thermocouples 10 degrees on each side of the load vector of each bearing were likewise displaced 180 degrees for the stationary and movable housings. The relationship of the oil jet and thermocouples to the direction and position of the load vector was the same for all four bearings.

Misalignment was accomplished by mounting a nonparallel spacer on either side of the bearing outer ring (Figure 6), forcing the outer ring to assume a tilted position with respect to the inner ring. The spacers were prevented from rotating by a tab loosely fitting in a matching slot to ensure the misalignment throughout each test. The bearings were physically symmetrical, so no particular orientation was required in assembly, although the serial number was always assembled to face the locknut. Thus, positions 1 and 3 were alike and 2 and 4 were alike.

The movable housings, although free to move in planes normal to the shaft, were held intimately against the inside face of the rig housing or cover by four cap screws appropriately spaced. The head of each screw was seated on a large flat washer, which in turn was seated against a concentric tubular spacer 0.0003 to 0.0005 inch longer than the thickness of the housing.

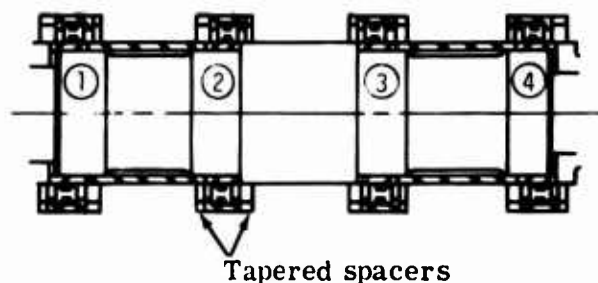


Figure 6. Misaligned Bearing Configurations.

Holes in the movable housing 25 percent larger than the tubular spacers provided adequate radial freedom but virtually no axial freedom. The angular misalignment introduced into the test bearings resulted entirely from the two tapered washers mounted on either side of the bearing outer ring, as shown in Figure 6.

TESTING

The test conditions were originally chosen to produce skidding and roller wear in about 10 hours (five cycles of 2 hours each; see Figure 7) of running on each bearing. Two sets of four bearings each were purchased to make the trial runs: one set to verify the conditions, and the second set to confirm the results of the first. These 8 bearings were not manufactured to the same selected levels of the dimensional variables as the test bearings, but to "standard" dimensions for a conventional bearing with the same envelope. The same measurements, however, that were taken for the test bearings were also taken for these, thus providing a complete geometrical background to aid in analyzing subsequent test results.

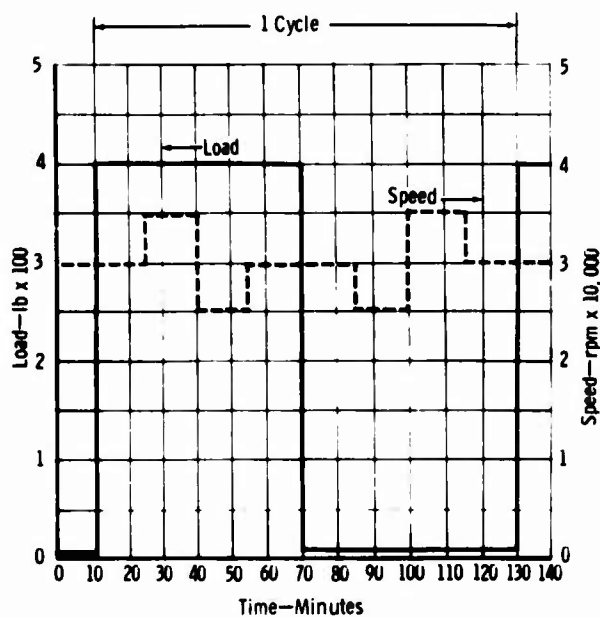


Figure 7. Bearing Test Schedule.

After the rig was assembled and the shakedown test was run with four spare, or slave, bearings, the rig was assembled with the verification bearings. MIL-L-7808F, Stauffer Type I, Lot 2718, lubricant was used throughout the program.

After one 2-hour cycle test, the bearings were inspected and found to be in good condition (there had been apprehension about the two misaligned bearings ---positions 2 and 4---even running under what was considered a severe misalignment condition), and therefore the apprehension was relieved. It was found however, that test rig vibration was a problem, particularly with a 400-lb load and at 35,000 rpm. After several vibration surveys and many corrections and adjustments, it was determined that the cause of the difficulty was a bent shaft between the motor and the Varidrive on which a cantilever gear was mounted.

During the original and the confirming set of trial bearing tests, the following events occurred, and two significant changes were made:

- After a few hours of running, the shaft and bearing bores were badly fretted, as a result of a combination of a nonhardened shaft surface, too little interference fit, and high vibration. The shaft was reground to approximately 0.004 inch undersize, followed by hard chromium plating and finish grinding to 1.5750 inch, which was 0.0003 inch larger than the size prior to the rework. This was considered to be the maximum allowable increase in shaft size without adversely affecting the bearings having the low level of internal radial clearance.
- Under the operating conditions originally specified, viz., 40 lb to 400 lb load per bearing, 25,000 to 35,000 rpm, and 1.3 lb of lubricant per minute per bearing, no significant evidence of either skidding or roller end wear had occurred. The conditions had to be made more severe in order to produce the damage response desired. Therefore, the low load was reduced to 10 lb, and the lubricant flow was reduced to 0.5 lb per minute per bearing. The response on the confirming set of bearings indicated that a satisfactory skid pattern had developed on No. 1 inner race. Although a satisfactory roller end wear response had not occurred on any of the four bearings, it was believed that the wide range between levels of the test parameters would induce wear on the roller ends; therefore, the operating conditions were considered to be acceptable. Subsequent running indicated that the right choice had been made with respect to both time and conditions. Table III shows the entire rig build schedule. Note the randomness of bearing selection and operating variables.

TABLE III. RIG BUILD SCHEDULE													
Build No.	Rig Position 1			Rig Position 2			Rig Position 3			Rig Position 4			No. of Teardowns & Assemblies
	A	B	C	A	B	C	A	B	C	A	B	C	
1*	4	1		12	1		11	1		8	1		5
2**	7	1		2	2	-	13	1		1	2	+	6
3**	3	1		9	2	-	6	1		5	2	+	1
4	61	2	-	18	2	+	67	1		49	1		3
5	26	2	-	14	2	-	33	2	-	48	2	+	3
6	25	1		71	2	-	41	2	-	52	1		3
7	43	2	+	59	1		57	1		31	2	+	5
8	62	2	-	19	1		77	2	+	53	2	-	4
9	64	1		38	1		60	1		66	2	-	1
10	68	2	+	21	1		36	1		69	2	+	2
11	65	1		35	1		55	1		27	2	+	1
12	76	1		74	1		32	1		28	1		1
13	50	1		75	2	+	17	2	+	39	2	-	1
14	16	2	+	15	1		20	1		22	1		1
15	30	1		47	2	-	46	1		45	2	-	2
16	72	1		42	1		23	1		70	2	+	1
17	58	2	+	63	2	-	37	1		40	2	-	1
18	54	1		29	2	-	56	2	+	44	2	+	3
19	24	1		73	2	-	51	1		34	2	+	3
20	89	2	-	86	1		84	1		85	1		4
21	92	2	+	82	1		91	2	-	87	1		4
22	83	1		80	1		90	2	+	81	1		5
23	78	2	+	88	2	+	79	1		93	2	-	4
<p>*4 standard unmeasured bearings used in first rig build and shakedown.</p> <p>**2 sets of standard measured bearings used in establishing test conditions.</p> <p>A—Last 1 or 2 digits of bearing serial number GG00XX.</p> <p>B—1 indicates aligned bearing, 2 indicates misaligned bearing.</p> <p>C—+ indicates misalignment follows load zone; - indicates misalignment precedes load zone. See Factor 2, Figure 2a.</p>													

During the running of the third set of test bearings (Build 6; see Table III), the bent shaft was discovered to be the main source of the low frequency vibration. Replacement of the shaft reduced vibration at the source by 65 percent. Later comparisons revealed no apparent difference in the running temperature, wear patterns, etc., between bearings run before and after the shaft replacement.

After mounting each set of four bearings and checking the lubrication system for proper temperature and flow, the rig was started under a 200-lb-load and 15,000 rpm until the outer ring temperature reached a

point of equilibrium; then, without shutdown, the first cycle was started, i. e., 400-lb load at 30,000 rpm (see Figure 7). At the end of each 15-minute period of a given load and speed condition, a complete reading was taken and recorded on the log sheets for the first cycle of the test. Figure 8 shows bearing temperatures during a typical test cycle. A complete reading of data was taken generally at the beginning, middle, and end of a cycle. A complete reading consisted of data shown in Tables IV and V.

During the early running, several failures of test bearings occurred in which all evidence of initial distress was destroyed. In order to reduce this type of failure, the test rig was modified from an intermittent outer race temperature control to a continuous temperature controlled shut down device. This provided cyclic monitoring between bearings 1 and 2 and bearings 3 and 4 every few seconds. This method proved to be highly reliable and reduced bearing seizure failure from 6 out of 18 to 2 out of 48.

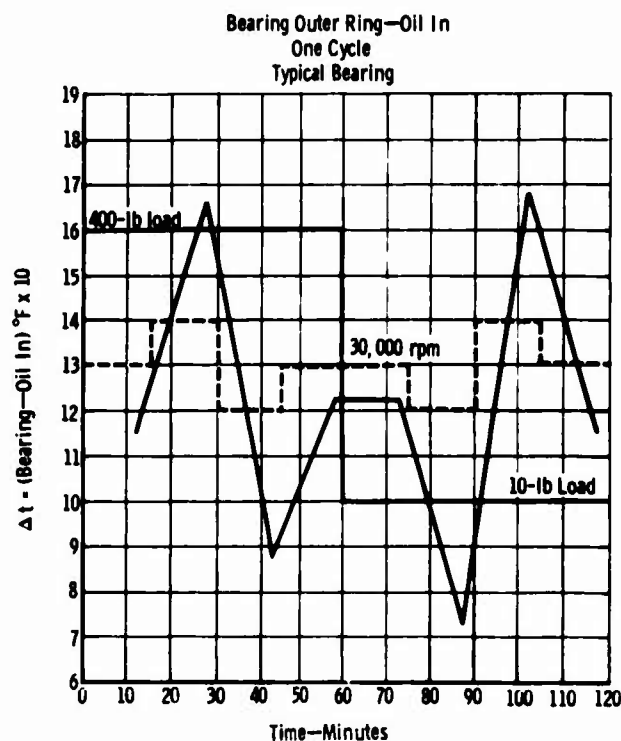


Figure 8. Temperature Rise.

TABLE IV. TEST DATA																			
Clock Time	Test Points Comp.	G/B input, (rpm)	Pressures (psig)										Pump Disch	Vibration				Axial	
			Oil supply							Ball Brg	G/B	Displ		Velocity					
			Load Cyl	Brg No. 1	Brg No. 2	Brg No. 3	Brg No. 4	Mils	Frequency (cpm)			In./Sec		Frequency					
2200		Start drive motor																	
2205		640	18	9	9.5	8	8	8	8	58	450	Horizontal	1.0	300/900	2/5	200/1000	2/4	300/700	
2210		724	180	8	9	7.5	7.5	7.5	8	57	450	1.0	800/1200	0.25	0/500	0.5	1000/1600		
2215		723	180	8.5	9	8	8	8	8.5	56	450	0.75	1200/3000	0.2	1200/3500	0.8	1200/1800		
2220		722	180	8.5	9	8	8	8	8.5	56	450	0.8	1400/2800	0.5	300/1100	0.8	2200/3000		
2230		Brought to point 1a endurance conditions																	
2245	1a	1449	350	9	9.5	8	8	8	9	52	450	6.2	1250	4/8	200/500	0.9	1900/2300		
2300	1b	1692	350	8.5	9	8	8	8	8.5	49	450	2.6	500/600	2.5/6.5	300/700	1.8	1900/2100		
2315	1c	1210	350	9	9.5	8	8	8	9	47	450	2.3	1200/2300	2/5	100/600	0.7	1200/2200		
2330	1d	1451	350	9	9.5	8	8	8	9	45	450	5.2	1150	6	50/500	1.5	1800		
2345	1e	1451	18	9	9.5	8.5	8.5	8.5	9	46	450	5	1800/2100	3/6	200/1000	1.3	1200/2100		
2400	1f	1209	18	9	9	8	8	8	8.5	46	450	2.3	1800/3000	2/5	200/2000	1.0	1000/2000		
0015	1g	1691	18	9	9.5	8	8	8	9	45	450	2.7	2000/3000	5/7	200/600	1.2	900/2000		
0030	1h	1450	18	9	9.5	8	8	8	9	44	450	6.1	1400/2400	5/7	200/650	1.0	2500		
0045	2a	1450	350																
	2b	1640	350																
Motor trans & rig all shut down during change of speed from point 2a-2b at 0345 hr.																			

TABLE V. TEST DATA																
Clock Time	Rig Oil In No. 1	Temperatures (°F)										Oil Out				
		Test Bearing Outer Race														
		2 No. 1	3 No. 1	4 No. 2	5 No. 2	6 No. 3	7 No. 3	8 No. 4	9 No. 4	10 No. 3	11 No. 4	12 No. 5				
2200	Start drive motor															
2205	185	179	180	169	169	169	174	154	158	133	138	156				0.23
2210	178	188	190	188	187	183	188	166	170	178	186	168				0.29
2215	178	199	199	200	199	192	197	175	179	182	193	177				0.26
2220	181	208	206	206	206	199	201	181	187	187	197	183				0.23
2230	Brought to point la endurance conditions															
2245	172	292	287	296	289	282	292	256	254	237	253	237				0.65
2300	175	339	339	342	335	330	342	301	310	265	292	275				0.67
2315	184	291	288	301	293	286	293	261	262	244	257	259				0.46
2330	175	309	306	312	305	296	305	271	278	250	268	260				0.53
2345	177	293	289	298	293	293	303	270	275	244	260	261				0.45
2400	174	253	251	261	257	264	270	257	250	232	233	243				0.33
0015	180	354	355	336	332	339	350	305	312	269	297	282				0.37
0030	181	300	298	308	302	302	312	280	283	252	268	272				0.50
0045	0.184	324	322	324	318	310	322	281	286	255	282	272				

Except for running-time response, seizure failures contributed no information in this investigation. The bearings were quite vulnerable to seizure for two reasons: the operating conditions had to be severe to produce the damage response desired with low operating time, and the low limit of clearance had to be small so that with a significant spread in the two levels, the high level would still be realistic. However, some thermal growth took place and in some instances all the radial clearance was lost. All the test bearings were stabilized, and the temperature sensing cutoff was approximately 400°F at the outer ring. The presence of some growth was expected.

Whenever a bearing became distressed to the point of requiring shutdown, it was replaced by one of the slave bearings used in the original buildup until the surviving test bearings had completed their 10-hour run.

After completion of the test on each set of four bearings, they were removed, preserved in oil, and packaged in the original carton. Great care was taken to preserve the same relationship among the components; i.e., each roller was kept in the cage pocket in which it ran and was oriented in the same direction. Then, by careful examination and study, wear patterns and damage were meaningful.

During examination, each bearing was carefully disassembled, and each component was placed in a special holder to maintain its relation to every other component. Every surface was inspected under magnification and charted. Other data believed to be useful, such as dimensions, and surface finishes, were also recorded. The end wear of the rollers was measured on a Proficorder with a 0.0001-inch-radius stylus. If the end wear in a given bearing was generally uniform among the rollers, only sample measurements were taken.

Following the compilation of all inspection data, descriptors were searched for that could be used in the scientific analysis of the test. The following conditions were rated: measurement of roller end wear, running time, skidding, wear, skidding and wear, fretting on bearing bore, outer ring temperature rise above oil in temperature, and general condition.

## ANALYSIS

The geometric and operational factors considered were those which most likely influence end wear and skidding. Plane misalignment was a factor only if the bearing was misaligned. Therefore, two separate analyses were performed and reported: the misaligned and the aligned. Skidding, as affected by the seven factors, has been examined, but the general mathematical treatment could not be applied because of an insufficient number of specimens showing the condition.

The significant factors in the two analyses, in descending order of importance, are:

<u>Misaligned Bearings</u>	<u>Aligned Bearings</u>
Roller end clearance	Cage geometry
Cage geometry	Crown
Internal clearance	Roller end clearance
	Internal clearance

To construct a valid half-replicate experiment, it is assumed that at least one factor or interaction has no effect upon the dependent variable, end wear. In the  $1/2 \times 2^7$  factorial experiment, it was assumed that the seven-factor interaction had zero effect on end wear. Denoting the experimental factors by the letters A through G, the half-replicate experiment was constructed by using the ABCDEFG interaction as the defining contrast, where A is the code for crown, B for roller end clearance, C for internal clearance, D for flange angle, E for cage geometry, F for misalignment, and G for plane of misalignment. The defining contrast, in statistical parlance, is that factor which contains no information regarding the dependent variable(s). The alias system that emerged is:

I	- ABCDEFG	ABD	- CEFG	BCD	- ADFG
A	- BCDEFG	CD	- ABEFG	ABCE	- DFG
B	- ACDEFG	ACD	- BEFG	DE	- ABCFG
AB	- CDEFG	BCD	- AEFG	ADE	- BCFG
C	- ABDEFG	ABCD	- EFG	BDE	- ACFG
AC	- BDEFG	E	- ABCDFG	ABDE	- CFG
BC	- ADEFG	AE	- BCDFG	CDE	- ABFG
ABC	- DEFG	BE	- ACDFG	ACDE	- BFG
D	- ABCEFG	ABE	- CDFG	BCDE	- AFG
AD	- BCEFG	CE	- ABDFG	ABCDE	- FG
BD	- ACEFG	ACE	- BDFG		

The alias system exists because only half the information contained in the complete experiment is present in the half replicate. The aliased pairs are algebraically equal.

In analyzing the test data, significant main factors (aliased with six-factor interactions) and two-factor interactions (aliased with five-factor interactions) were presumed to be end wear associated; this assumption is valid in that higher order interactions, in the general sense, are nonsignificant. Three-factor interactions and their four-factor aliases were not included in the analysis.

In this sequence, Stage 1 testing was defined and executed, and the data were analyzed to determine significant factors. Stage 2 testing was defined and executed, and all test data were analyzed. Stage 2 testing was defined by fifteen additional tests to define the nonlinear effects associated with those test variables determined to be significant in the Phase I program.

In executing the test schedule (both stages), four bearings were mounted along a single shaft. The load on the outer bearings was directed downward, and the load on the center two was directed upward. Small differences in loading, or differences in oil flow or temperature to the bearings, could result in wear differences that would be attributed to the experimental factors. To preclude this type of erroneous conclusion, bearings were positioned at random within the test apparatus. Position differences were eliminated by the averaging process used to evaluate experimental variables.

Roller end wear was the numerical quantity analyzed. Wear was measured on the near side and the far side of the rollers. The bearing serial numbers were stamped on the near side; in all cases lubrication was applied to the far side. Separate analyses were performed on near-side data, far-side data, and the two-side average under the two categories of misaligned and aligned (6 analyses). The two-side average was considered to be most indicative of test variable response and was used as the basis for conclusions.

The specific results of those analyses are presented in this section. These results are stated in terms of equations that describe, or predict, end wear as a function of the significant test variables. A "t" value is associated with the factors or factor products that are shown; the "t" shown is the Student's "t", but may be regarded as an importance factor in the sense that larger "t" values denote a closer association between the test variables and end wear.

All equations have the form

$$\text{Wear} = \text{EXP} (B_0 + B_1 X_1 + B_2 X_2 + . . . + B_k X_k)$$

For notational convenience, the coefficients, the associated "t" values, and the geometric factors denoted by  $X_1, X_2, . . . , X_k$  are presented in Tables VI and VII.

Using Tables II and III, any equation can be mapped into its linear form,  $Y = \text{EXP}(B_0 + B_1 X_1 + B_2 X_2 + . . . + B_k X_k)$ , by proper substitution. The linearized version of the misaligned bearing two-side average equation is:

$$\text{Wear} = \text{EXP} \left\{ 0.1554 + 0.0712 (\text{roller end clearance}) - 0.0002 (\text{roller end clearance})^2 \times (\text{cage geometry}) + 0.000044 (\text{roller end clearance}) \times (\text{internal clearance})^2 \right\}$$

Wear is computed in ten thousandth of an inch, using input in the units indicated below:

Degree misalignment (minutes)  
 Plane of misalignment (degrees)  
 Degree crown (inches  $\times 10^4$ )  
 Roller end clearance (inches  $\times 10^4$ )  
 Internal clearance (inches  $\times 10^4$ )  
 Flange angle (minutes)  
 Cage geometry (inches  $\times 10^3$ )

The "t" values provide a comparative index to measure the strength of the relationship between wear and the corresponding test variable. Large "t" values denote a close association. Alternately, "t" values provide a measure of confidence that a true relationship exists between dependent and independent variables. The confidence associated with any specified "t" value (given degrees of freedom) is determined by the area associated with the magnitude of the "t" value. Several "t" values and their associated confidences are as follows:

<u>"t"</u>	<u>Confidence (DF* = 60)</u>
3.460	0.999
2.660	0.99
2.000	0.95
1.671	0.90
1.296	0.80

\*Degrees of Freedom

TABLE VI. ALIGNED BEARING

NEAR-SIDE WEAR		
Coefficient	't''	Experimental Factor
0.4963	*	$\beta_o = \text{constant}$
-0.0585	5.1	$(\text{Degree crown}) \times (\text{Cage geometry})^2$
-0.0012	4.0	$(\text{Degree crown}) \times (\text{Roller end clearance})^2$
0.0043	2.0	$(\text{Roller end clearance}) \times (\text{Internal clearance})$
0.0905	1.9	$(\text{Cage geometry})$
0.0002	1.8	$(\text{Degree crown}) \times (\text{Flange angle})^2$
-0.0001	1.7	$(\text{Roller end clearance})^2 \times (\text{Internal clearance})$
<u>FAR-SIDE WEAR</u>		
0.3131	*	$\beta_o = \text{constant}$
0.0506	4.9	$(\text{Roller end clearance})$
-0.0477	4.2	$(\text{Degree crown}) \times (\text{Cage geometry})^2$
0.0023	4.1	$(\text{Degree crown}) \times (\text{Internal clearance})^2$
0.2365	3.8	$(\text{Cage geometry})$
0.0009	2.8	$(\text{Degree crown}) \times (\text{Roller end clearance})$
-0.0002	2.3	$(\text{Roller end clearance}) \times (\text{Cage geometry})$
-0.6849	2.3	$(\text{Degree crown})^2$
-0.0001	2.2	$(\text{Internal clearance})^2 \times (\text{Flange angle})$
0.0016	1.8	$(\text{Internal clearance}) \times (\text{Flange angle})$
-0.0002	1.6	$(\text{Internal clearance})^2 \times (\text{Cage geometry})$
<u>TWO-SIDE AVERAGE</u>		
0.3908	*	$\beta_o = \text{Constant}$
-0.0543	5.2	$(\text{Degree crown}) \times (\text{Cage geometry})^2$
0.0364	3.2	$(\text{Roller end clearance})$
0.1671	3.1	$(\text{Cage geometry})$
0.0009	3.0	$(\text{Degree crown})$
-0.0002	2.0	$(\text{Roller end clearance})^2 \times (\text{Cage geometry})$
0.0008	1.8	$(\text{Degree crown}) \times (\text{Internal clearance})^2$
*Not computed for the constant		

TABLE VII. MISALIGNED BEARING (10 MINUTES)		
Coefficient	"t"	NEAR-SIDE WEAR
		Experimental Factor
0.1542	*	$\beta_o = \text{Constant}$
0.0748	6.1	(Roller end clearance)
-0.0002	2.3	(Roller end clearance) <sup>2</sup> × (Cage geometry)
0.000046	2.1	(Roller end clearance) × (Internal clearance) <sup>2</sup>
-0.00000735	1.7	(Roller end clearance) × (Flange angle) <sup>2</sup>
FAR-SIDE WEAR		
-0.1565	*	$\beta_o = \text{Constant}$
0.0879	7.8	(Roller end clearance)
-0.0002	3.4	(Roller end clearance) <sup>2</sup> × (Cage geometry)
0.0010	2.1	(Internal clearance) <sup>2</sup>
0.0007	1.5	(Plane misalignment) × (Cage geometry)
TWO-SIDE AVERAGE		
0.1554	*	$\beta_o = \text{Constant}$
0.0712	6.3	(Roller end clearance)
-0.0002	3.1	(Roller end clearance) <sup>2</sup> × (Cage geometry)
0.000044	2.0	(Roller end clearance) × (Internal clearance) <sup>2</sup>
*Not computed for the constant.		

The following listing shows end wear comparison among the categories that are relevant to this analysis.

	Wear in Inches × 10 <sup>4</sup>	
	<u>Aligned</u>	<u>Misaligned</u>
Near side	3.0838	4.1452
Far side	3.9012	4.3712
Average of two sides	3.5824	4.3656

Several bearings failed in the catastrophic sense. These bearings were completely destroyed, and no information was available regarding end wear at time of failure.

The values assigned to the experimental variables are shown in Table VIII.

TABLE VIII. VALUES ASSIGNED TO EXPERIMENTAL VARIABLES ASSOCIATED WITH CATASTROPHIC FAILURE						
Serial Number	Degree Misalignment (minutes)	Plane Misalignment (degrees)	Decree Crown (inch)	Roller End Clearance (inch)	Internal Clearance (inch)	Flange Angle (minutes)
<u>Misaligned bearings</u>						
78	10	67.5	0.0001	0.0007	0.0007	0
68	10	67.5	0.0001	0.0007	0.0007	0
34	10	67.5	0	0.0030	0.0007	0
31	10	67.5	0	0.0007	0.0007	0
48	10	67.5	0.0001	0.0030	0.0007	0
43	10	67.5	0	0.0030	0.0007	45
61	10	-67.5	0.0001	0.0030	0.0007	45
<u>Aligned bearings</u>						
24			0	0.0030	0.0007	45
25			0	0.0030	0.0007	0
52			0.0001	0.0007	0.0007	45
86			0.00005	0.00185	0.0025	27.5
						0.005
						0.005
						0
						0.003

A summary of relevant facts that emerged from the data analysis follows:

- The wear response as a function of the test variables was greatly different for misaligned and aligned bearings, and to a lesser degree was different for near-side and far-side wear.

Seven misaligned bearings failed, 4 aligned bearings failed.

- Catastrophic failures for the misaligned bearings were primarily associated with the positive plane of misalignment.

In the misaligned category, the plane of misalignment was positive in 6 of 7 bearings, 5 of 7 had zero flange angle, and all had an internal clearance of 0.0007 inch, which was the minimum value defined.

- Average wear for misaligned bearings was greater than for the aligned bearings.
- The fact that all except one failed bearing had the minimum internal clearance of 0.0007 inch is significant.

It is presumed that the misalignment was the most significant variable and that the other test variables affect end wear to a lesser degree. This is substantiated by the fact that more test variables were significant in the aligned situation; also, a higher wear rate was observed in the misaligned bearings. Differences in near-side and far-side wear for these bearings (that were otherwise subjected to the same experimental conditions) can be explained by the amount of lubrication that reached the wear point; consider that oil was applied to the far side only and that lube differences existed.

Inner ring stabilization can perhaps explain the predominance of failures at 0.0007 inch internal clearance. Apparently inner ring growth was experienced to the extent that some internal clearances may actually have been negative.

Test data were analyzed using the 360-65 computer system operating with the stepwise regression program OSBB38. This program uses the Least Squares technique to select factors that associate with end wear and to compute parameters for the mathematical model:

$$\text{LN}(Y) = B_0 + B_1 X_1 + B_2 X_2 + \dots + B_k X_k.$$

In the overview of the testing, the analysis, and the results, it appears that an aligned bearing with sufficient dimensional clearance to prevent catastrophic failure presents the optimum operating situation. If stresses are placed upon the bearing, as in the misaligned situation, wear can be minimized by increasing some other clearance to provide stress relief, as with increased roller end clearance.

This testing and subsequent data analysis have demonstrated the feasibility of using statistical methods to evaluate and more fully understand the complex interrelationships that exist between wear and the combination of operational and geometric factors that go into bearing design and operation.

Although skidding could not be analyzed by the same mathematical treatment, some observations follow:

Fifteen bearings had evidence of skidding on the inner race---the surface where it invariably appears first. Three were severely damaged (see Figures 9, 10, and 11); the remaining twelve had only slight evidence of skid damage. Thirteen had completed the 10-hour test with the two exceptions running 3.21 and 2.14 hours. These two bearings, which had only slight skid damage, were assumed to have gotten worse by the time 10 hours were accumulated; they were placed in the same category with the three severely skidded bearings. Two groups were formed for comparison. Serially numbered bearings GG0051, 57, 59, 65, and 69 were placed in the severely skidded group; GG0017, 23, 27, 28, 40, 45, 46, 58, 70, and 72 were placed in the slightly skidded group. Comparing on the basis of the seven 2-level factors (see Table IX), there appears to be no well-defined correlation.

Generally, the number of occurrences at the two levels are reasonably random.

Other factors that may have influenced the test results were:

- Finish---The manufactured finish was 4 or 5 microinches AA (Arithmetic Average) on the inner races and a constant value of 2 microinches AA on the rollers.
- Out-of-Round---The average out-of-round inner race before mounting was  $65.7 \times 10^{-6}$  in., varying between a maximum of  $100 \times 10^{-6}$  and a minimum of  $45 \times 10^{-6}$ . The average out-of-round outer race before mounting was  $90 \times 10^{-6}$ , varying between a maximum of  $160 \times 10^{-6}$  and a minimum of  $60 \times 10^{-6}$ .

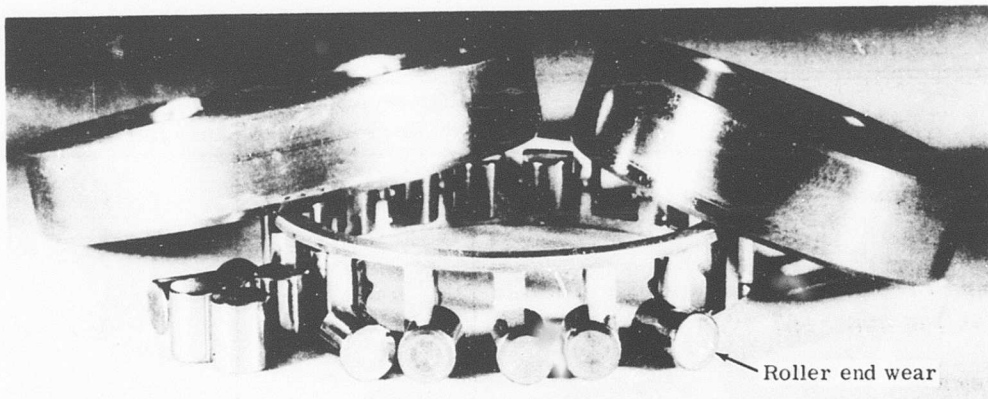


Figure 9. Roller End Wear.

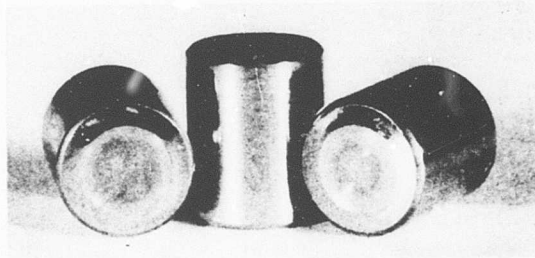


Figure 10. Roller End Wear (2X).

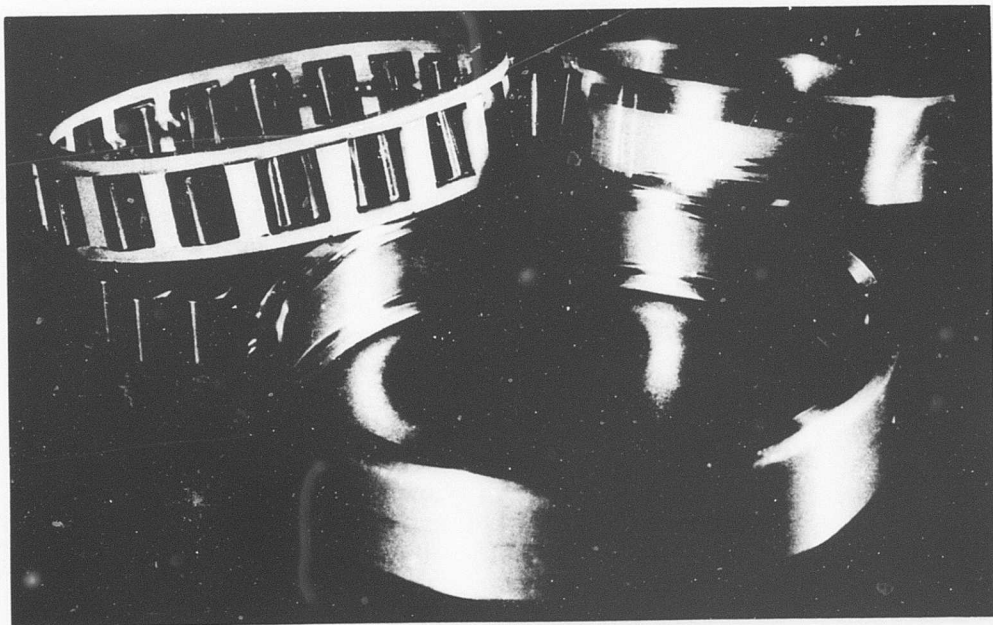


Figure 11. Skidding Damage on Inner Race.

TABLE IX. COMPARISON OF TWO-LEVEL FACTORS			
Factors		Fifteen Skidded Bearings	Five Severely Skidded Bearings
Misalignment	10 minutes	7	1
	0 minutes	8	4
Plane of Misalignment	+67 degree 30 minutes	6	1
	-67 degrees 30 minutes	1	0
Crown	0	6	0
	0.0001 inch	9	5
Roller End Clearance	0.0030 inch	7	3
	0.0007 inch	8	2
Internal Clearance	0.0022 inch	10	3
	0.0007 inch	5	2
Layback Angle	0	4	1
	45 minutes	11	4
Cage Pocket	0	8	2
Skew	0.005 inch	7	3

- Position in Test Rig—The number of skidding occurrences in the four rig positions were: 3 in position 1, 1 in position 2, 5 in position 3, and 6 in position 4, or 6 in the movable housings and 9 in the stationary housings.
- Builds ---The distribution among the rig builds was: 1 occurrence in 6 builds or 24 bearings, 2 in 4 builds, and 3 in 1 build.

None of these observations indicates other than random occurrences. It is therefore concluded that neither the bearings, the test rig, nor the conditions of operation were significant with respect to skidding. During the preliminary running to produce the desired response and test duration, reduction of oil flow from 1.3 to 0.5 pound per minute per bearing was required to produce skidding. Besides the two well-known causes of skidding (high speed and light load), oil flow may be another. Except to the extent that bearing size affects skidding, within the limitations of this program none of the seven factors selected for end wear is significant with respect to skidding.

## CONCLUSIONS

Roller end wear was analyzed carefully by the mathematical techniques usually applied through design experimentation and statistical analysis. The serially numbered side opposite from which the lubricant was introduced (near side), the side from which the lubricant was introduced (far side), and the two-side average were considered in separate analyses. The general conclusions that follow are based on consideration of the two-side average.

### MISALIGNED CONDITION

- A low value of roller end clearance is required to reduce roller end wear.
- Roller pocket squareness is important in preventing roller end wear. However, increased roller end clearance up to the nominal of the range of values tested is beneficial in offsetting the deleterious effects of pocket out-of-squareness. Moreover, if roller end clearance is increased to offset pocket out-of-squareness, internal clearance should be reduced to its minimum value.
- The other controlled variables in the experiment did not closely associate with roller end wear; therefore, they may vary within the range of values tested but preferably are controlled toward the mean.

### ALIGNED CONDITION

- Low values of roller end clearance and roller pocket out-of-squareness are required to reduce roller end wear.
- Roller pocket squareness is important and should be closely controlled. However, if some pocket out-of-squareness is present, some crown and roller end clearance aids in reducing end wear, roller end clearance being more sensitive than crown. Crown, as used in this experiment, should not be confused with its primary function of relieving concentrated loads at the roller ends.
- Low value of internal clearance is beneficial in the presence of crown in reducing roller end wear, and if crown is minimum roller end clearance should be less deleterious.
- Roller end wear is invariant with respect to flange angle.

Since these conclusions are based on the particular size and design of bearings used in the experiment under specific operating conditions, care should be exercised in applying them.

### RECOMMENDATIONS

Based on the results of this program, the following specific values for the variables tested may be established for this or a similar size roller bearing operating under high-load and -speed conditions.

- Roller end clearance---0.0005 to 0.0010 inch.
- Cage pocket squareness---0.001 inch maximum. This should be controlled on the drawing.
- Internal clearance after assembly on shaft---0.0005 to 0.0015 inch with consideration given to operating temperature differential between the inner and outer rings.
- Crown---0.00005 inch is desirable for end wear but it must also be compatible with load. Except for very light loads, 0.0001 inch would be more practical.
- Flange angle---10 to 30 minutes.
- Although misalignment is always undesirable, it should not exceed 0.001 inch per inch of diameter - 35 percent of the value tested which is within good aircraft manufacturing practice.

The above recommendations may be applied only as guidelines for other bearing sizes and conditions of operation.

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13. ABSTRACT An experimental study to evaluate the effect of five geometry factors and two operational factors on roller end wear and skidding in roller bearings is described. Running each of 80 test specimens for five two-hour cycles or until failure constituted the physical testing which was based on a one-half replicate of a 2 <sup>7</sup> factorial experiment. Results, shown in aligned and misaligned bearing operation classifications, indicate that end wear is affected to a greater extent by factor interactions than by main effects. No correlation is shown between the seven factors and skidding. Recommendations are made for the optimum values of the test variables to be used in roller bearings of the size, type, and under similar conditions of operation that will have the least probability of roller end wear.		

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